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University Design Challenge

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14. ABSTRACT
ASU has participated in AFOSR University Engineering Design Challenge for last 3 years. ASU team has designed, built prototypes and participated in three competitions. The objective of the university design challenge was to develop practical solutions to mechanical engineering problems that were critical to the DoD. The potential solution(s) were proposed and developed by the senior undergraduate students as a capstone project.

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University Engineering Design Challenge

Final Report

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Grant FA9550-11-1-0345, "University Design Challenge,"

Contents

Executive Summary:

ASU has participated in AFOSR University Engineering Design Challenge for last 3 years. ASU team has designed, built prototypes and participated in three competitions. The objective of the university design challenge was to develop practical solutions to mechanical engineering problems that were critical to the DoD. The potential solution(s) were proposed and developed by the senior undergraduate students as a capstone project. The students decided the project milestones and were responsible to meet them in a timely fashion. The faculty mentor was responsible to ensure that the students have the resources they need and are on track. In general, the approach was based on 'plan your work and workout your plan' philosophy and comprised of following steps. 1. Understanding the problem 2. Defining the goals/deliverables 3. Identification of constraints and resources 4. Design Process 5. Brainstorming and Design Matrices and Subsystem Design/ Selection 6.Comprehensive Design Review (CDR) 7.Building the prototype/ Developing code 8. Integration, Testing and prototype refinement 9.End Product Delivery/Competition. In 2011-12 the team built a suction wall climbing device, in 2012-13 the team built a collapsible bridge and in 2013-14, the team built a heavy lift kit.

University Design Challenge 2011-2012:

In 2011-12, the ASU team developed a list of potential target systems and determined which of the climbing systems would be best, based on given requirements. After narrowing down the solutions, the team determined that our final product would use suction cups as an adhesion method, which would be distributed over four different pads attached to every limb of the soldier. It would also be flexible by letting the soldier maneuver and use his weapon in case of combat. Using all of the information gathered, the team designed a prototype using CAD. This prototype was subject to extensive research in relation to materials, weight, processes, and costs during the designing process. The prototype was thoroughly tested to ensure its functionality. Upon testing the prototype, the team identified important aspects of the design that required modification. Based on testing, modeling, and calculations, we developed the concept for our final design. This final design was evaluated for ease of use, compatible materials, and safety. Our product is economically efficient to build at an estimated price of $1,500. Throughout this project our information has proven that our product will do well in the market and be taken into consideration for use in the Air Force. The most important needs were the safety of the climber and ease of use, which freed the user to do other tasks. Time was a big concern for this project especially if we had a powered system with a limited battery life.

Suction Cups

The idea of having suction pads as an adhesion method opened a wide range of possibilities in the climbing design. In the beginning, this method seemed unrealistic due to the limitations of surfaces that a suction pad can operate on. The team found a lot of suction pad manufacturers in our research. We narrowed down the vacuum pads by styles and manufactures. We eliminated some competitors due to compressed air flow limitations. Considering cost and max loads of suction cups/pads we found a supplier (Wood’s Powr-Grip) that guaranteed a suction pad that can work on multiple surfaces, withstand a load greater than 300 lbs, and stay within our budget (as shown in Figure 1).

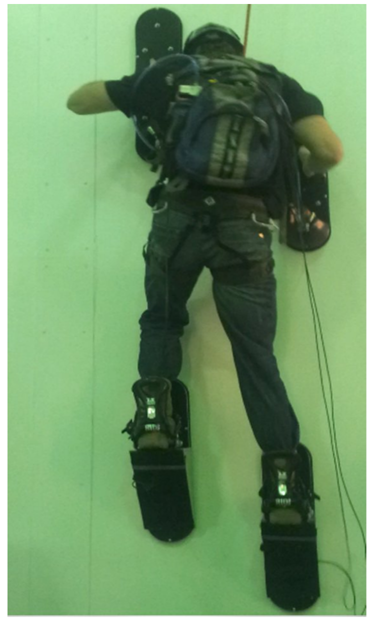


Figure 1: Suction cup system

Figure 2: Suction cup adhesion method testing

Upon testing our system (Figure 2) we encountered a few problems. We saw that we needed a larger pressure pump to control adhesion and hold greater loads. We tested suction cups on different surface materials while applying a load. These suction cups were able to hold loads when placed vertically. After successfully testing our system, we were able to make proper modifications that have made it more reliable and safe. Some of the pieces of equipment that needed to be included with these suction pads were:

1034X Dynaflo pump

12.8 V 10 Ah (Lithium Iron Phosphate) Battery Pack (Hand pads)

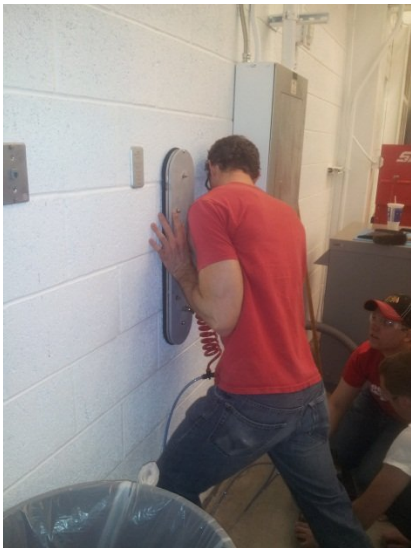
Secure, one-size-fits-all binding

Humphrey 310/410 3 way 2 position solenoid

Push button

Easy grip handle

Based off our preliminary design concept, we decided to prototype the hand and foot pads for our device. Before we started to make the prototype we further researched air flow, mathematical modeling, and edited our preliminary design. One of the first changes made to our preliminary design was the position of the suction pad, which we ended up making vertical. We did this because it was determined through testing that the suction pad would not hold as well when placed horizontally on the wall (figures 4 and 5).



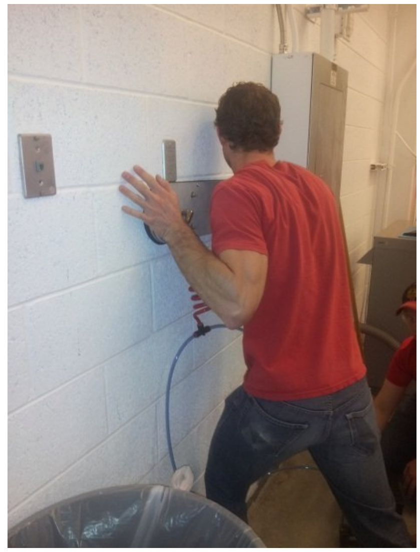


Figure 4: Testing pad Vertically **Figure 5: Testing pad Horizontally**

Next, it was important to decide where exactly to place the equipment that would power and run the suction pad system. After much thought, we decided to put the equipment in a custom made compartment, which would be attached to each suction pad, and in a backpack. This was done because we wanted the system to be organized and we did not want to have equipment and wires getting in the way. This was extremely important because we needed to make the system flexible and clean. We decided that both foot suction pads would have compartments and that for the hand suction pads, the equipment would be located on the backpack of the user. Since the hand suction pads were going to have handles with a button to power the system, placing the equipment into a backpack was the most reasonable thing to do. We also eliminated one battery and one pump from the hand suction pads. Although it slows the flow, through calculations we decided that one pump and one battery is enough to power and hold the upper portion of the system. The system subcomponents are shown in figures 6,7 8 and 9.

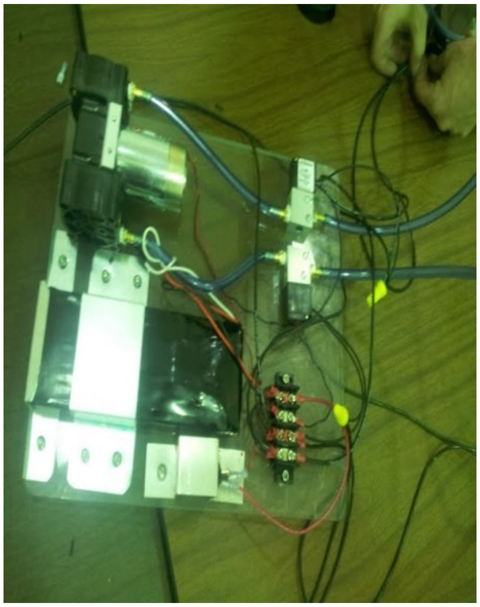


Figure 6: Hand Pad equipment board

Figure 7: Foot compartments

Figure 8: Hand Pads

Figure 9: The whole system w/backpack

Mathematical Modeling/ Engineering Analysis

Now that we had a general idea of our model, we tried to use mathematical models to determine theoretical information about our prototype. The first theoretical value that we found was the pressure created by our prototype. The equation that was used to find pressure is $P = F / A$ where *P* equals the pressure in (psi), F is the force (lbs) of the object applied, and A (in^2) is the reference area. Using this, the calculated force (weight) is 300lbs. The reference area of our device is ≈ 131 in^2. Calculating this we see that we need at least 2.3 psi from pump on suction cup to hold 300lbs. After looking online, it was determined that the pump's max pressure is 17 psi. This pump falls within our psi range. We did not know the amps needed so we got an amps vs. pressure plot to the amount of power needed. The resulting plot can be seen in (Figure 10). With this graph we got a rough estimate of the number of amps needed per psi that the pump gives off our device.

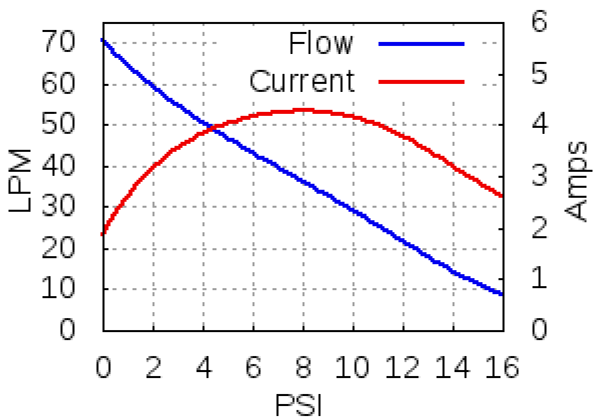


Figure 10: Amps vs PSI (image from Dynaflopumps.com)

Equations: T= time is takes user to climb 90ft

P= power needed

$$= 90 \quad \frac{1.26}{18} \quad \frac{1h}{60} = .105\ h$$

$$= 4.5 \quad 0.105h \ = .472 \quad\quad h$$

The pump was chosen based on the size and power needed for our system. We needed a lightweight, small, high flow (LPM), and low amp powered pump to give us enough suction for the pad. At just 2.5 lbs, this pump is an ideal choice for demanding applications requiring a compact design. This pump was perfect in size, weight, and power. The battery packs for both hand and foot suction pads were also perfect in size and in power. Since our desired pump worked of a current of 4.5 amps, we needed to consider a battery for each pump that would have our system working throughout the climbing time. During testing we determined that the user climbed 18ft in 1.26 minutes. That gives us an approximate time of climbing a 90ft wall in .105 hours. With that being said, the amount of power needed during the climbing time to power our pumps is .472 amp-hours. The T-energy rechargeable battery was be a good choice for this system. It is a 12.8V Lithium Iron Phosphate battery with a 10 amp-hour rated capacity, which is more than enough to power our system throughout the climb. After testing we determined that the battery had the ability to hold a charge for over three hours when the pump is operating at maximum load. The solenoid 3 way 2 position was chosen due to the necessity of a port that can provide atmospheric pressure for the release of the suction.

Foot Bracket Stress Analysis

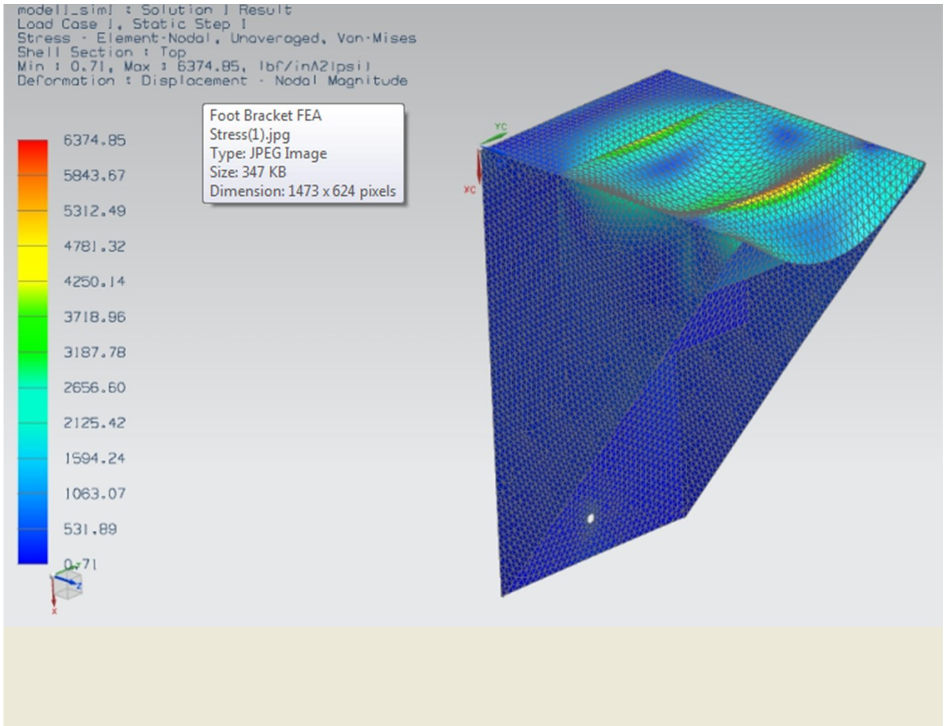


Figure 11: Foot Bracket Stress Analysis

The team did a foot bracket stress analysis as shown in figure 11 to see how much load our desired material can handle before deformation. This analysis shows the max load stress to be 6375 psi. The yield strength of aluminum 6061 T6 is 40,000 psi. That means that there is a factor of safety of 6.27 before there is any permanent deformation.

Safety

Risk Assessment Matrix

In order for us to be successful in our project, we needed to make safety our number one priority. We were able to identify the hazards and risks within the project to minimize potential injury. We made a risk control plan as a guideline to assist the safety review board in addressing the hazards with the risk management processes. Additionally we made a climbing safety checklist-Risk assessment as shown in Table 1 which we will adhere to before climbing the wall to ensure the climber's safety.

Conclusion

Our final design had great potential to be integrated into use by the Air Force. It was lightweight, compact, easy to use, and safe for the user. Our testing showed that this system could climb a minimum of 90ft. Also this system can be used in different environments either for a covert mission, a rescue mission, or simply anywhere a user needs to climb. This system was unique and well thought out, and, given additional modifications, might become a technological advancement for the future.

Table 1: Risk Assessment/ Control Plan

Risk Assessment					
Task & Hazards	**Person affected & location**	**Risk Rating**	**Risk Control Measures**	**By who**	**Notes**
Task: Lifting 6 lb. Suction Pads/ 5lb Nail gun **Hazards:** Back or shoulder strain or sprain	Climber Testing site	B4 LOW	**Currently:** Trained in good lifting technique Tested closed to lifting area Seek assistance if needed **Next Steps:** Add secure handles for ease in lifting Modify and replace suction grip material with light plastic material	KC	Suction cups/ Nail gun is subject to being attached to a linear actuator for less strain on climber
Task: Suction cup board adhesion fails/ Nail gun fails to attach **Hazards:** Climber falls to ground	Climber Testing site	C2 MEDIUM (Depending on height)	**Currently:** There will be 4 suction cups at all times in design **Next Steps:** Make sure to attach a bracket to the climbing surface to provide a anchor point for the climber to be secure in place at all times	Entire Team	Incase climbers suction cup fails, he/she will be secured with other suction cups in place since they have safety rating 4 incase one malfunctions
Task: Climber drops Suction Cup/ Nail gun **Hazards:** Injury to bystanders getting hit by falling object	Testers/ Teammates/ Testing site	C3 LOW	**Currently:** Incase a suction cup is dropped, climber will be secured with 3 suction cups in place **Next Steps:** Attaching Suction Cup/ Nail gun to the climber harness incase it drops it wont fall to ground	Entire Team	Incase climber drops a suction cup, they will reel in dropped suction cup since its attached to harness and attempt reattachment
Task: Battery/ Pump/ Electrical Failure **Hazards:** Climber falls to ground	Climber Testing site	C1 HIGH (Depending on height)	**Currently:** Equipment safety checklist to make sure all parts are functioning properly **Next Steps:** Have a parachute incase climber needs to drop for safety Make sure to attach a bracket to the climbing surface to provide a anchor point for the climber to be secure in place at all times	Entire Team	Incase climbers equipment fails, the climber will either drop for safety with parachute or wait until next climber succeeds the climb and get winched up
Task: Building surface structure failure (surface crumbles or breaks) **Hazards:** Climber falls to ground	Climber Testing site	C1 HIGH (Depending on height)	**Currently:** Climbing safety checklist to make sure climbing surface and conditions are met and approved There will be 4 suction cups at all times in design with safety rating 4 incase one malfunctions **Next Steps:** Make sure to attach a bracket to the climbing surface to provide a anchor point for the climber to be secure in place at all times	Entire Team	Incase climbing surface crumbles or breaks, the climber will reposition himself/herself and attempt reattachment with suction cup in new surface

University Design Challenge 2012 –2013

Introduction

Arizona State University (ASU) was contracted by the Air Force Research Laboratory (AFRL) for the 2012-2013 academic year to assist in generating a solution to navigation challenges that are frequently encountered by United States airmen. The solution was to take the form of a device that could function as both a ladder and a bridge, and would be subject to a number of constraints, including weight, length, and volume constraints. No “easy” answer to this problem existed and, as a result, a fair measure of innovation was employed while addressing it.

The Problem Statement

After reviewing AFRL’s design criteria, the ASU team summarized its project goals in the below statement. During the 2012-2013 academic year, the team worked to:

Engineer a multipurpose device that will allow for the innovative, safe, reliable and simple navigation of field gaps that are often encountered by military personnel.

The Solution

The team opted to pursue a device similar to a truss bridge. This solution is favorable primarily due to (1) its ability to collapse in a controlled fashion and (2) the impressive strength its members provide. Trusses are common load-bearing structures, and are found in many modern-day applications due to their simple, strong, and effective designs.

Our design was based on research on trusses and how they have been effectively used in the real world. The following were some key attributes to our design that we felt separate it from other ladders. (a) Scissor motion, (b) highly compactable along longitudinal axis, (c) ropes improve lateral stability and relieve slats of loading by supporting tension forces, (d) modular concept, and (e) compression members lock device into place and allow it to be used as a ladder. Having a set length in rope and in the folding compression members allows for quick deployments. One only needed to extend the bridge outward and lock in place with the bronze compression members. Once locked, the bridge may be placed at the starting point and lowered or pushed to the end point.

Bridge Components

Slats

Carbon fiber, high strength-to-weight ratio, 0-90 orientation lends strength in two orthogonal directions, more suitable for three-dimensional forces than unidirectional layups.

Rungs

0-90 carbon fiber wrapped balsa, balsa has high strength-to-weight ratio and is extremely light. Balsa is too weak to support high loads, but when paired with carbon fiber, it supports compression forces while the fibers support tension forces.

Compression Members

Two thin bronze plates are attached together at the ends. C-channel sleeve was placed over this connection at the midpoint of the sleeve and all three members are riveted together. Once deployed, the plates will unfold and the sleeve will prevent any further rotation about the pivot point. Members will be used in a stabilizing role.

Feet

Aluminum/nylon to reduce weight. C-channels with wide, deep walls, one hole located through both walls which allowed the outermost slat members of the bridge section to be attached, by nut and bolt. This attachment allows for free rotation of the feet to improve grip and stability on uneven surfaces.

Supporting Rods

Smaller diameter/higher grade all-thread to reduce weight and provide stability. Placed all-thread rods through the bottom section of the truss in locations where it would be most beneficial.

Rope

Marine rope, high strength-to-weight ratio, relieves carbon fiber slats of tension.

Hardware

Nylon material and smaller diameter/higher grade bolts to reduce weight.

Analysis and Testing

The selection of the final bridge design hinged upon both theoretical analysis and real-world experimental testing on prototypes. Our test data is detailed below :

<u>3/15/13 Testing</u>

Trial 1

Parameters:

- Amsteel grey rope (fed through the bottom and top horizontals, as well as through verticals)
- No all-thread rods at the bottom
- Bridge expanded to about 45 degrees
- Bridge length is 6' 8"
- 0.5" thick slats, last section is comprised of doubled up pultruded slats
- Reference dimension (distance from the ground to the top of the rope at the bottom horizontal): 2.75"

Load (lb)	Rope distance from floor (in)	Deflection (in)
20	2.750	0.000
45	2.688	0.063
70	2.625	0.125
95	2.563	0.188
120	2.500	0.250
145	2.438	0.313
170	2.375	0.375
195	2.250	0.500
220	2.188	0.563
245	2.125	0.625
270	2.000	0.750
295	1.875	0.875

Observations: The bridge was laterally unstable. The nuts/washers connections may be contributing to this.

Trial 2

Parameters:

- Amsteel grey rope (fed through the bottom and top horizontals, as well as through the verticals)

- No all-thread rods at the bottom
- Bridge expanded to about 56 degrees
- Bridge length is 5’ 3.5”
- 0.5” thick slats, last section is doubled up pultruded slats
- Reference dimension (distance from the ground to the top of the rope at the bottom horizontal): 2.88”

Load (lb)	Rope distance from floor (in)	Deflection (in)
20	2.810	0.060
45	2.750	0.130
70	2.750	0.130
95	2.690	0.190
120	2.630	0.250
145	2.560	0.310
170	2.500	0.380
195	2.440	0.440
220	2.380	0.500
245	2.310	0.560
270	2.250	0.630
295	2.130	0.750
320	2.000	0.880

Observations: The cross-pieces that are not epoxied (i.e. the all-thread pieces that are not epoxied into the balsa wood) seem to be causing issues with regards to lateral instability.

<u>Testing</u>

Parameters:

- Amsteel purple rope, fed normally (horizontals and verticals), but then looped back horizontally through 3 sections
- All-thread at bottom (at each end, then two rods in the center of the bridge)

- Bridge expanded to 52.5 degrees
- Bridge length is 5’ 9.5”
- Bridge weight is 16 pounds
- 3/8” thick slats
- Reference dimension (distance from the ground to the top of the rope at the bottom horizontal): 2.25”

Load (lb)	Rope distance from floor (in)	Deflection (in)
20	2.190	0.060
45	2.130	0.130
70	2.060	0.190
95	2.000	0.250
120	1.940	0.310
145	1.880	0.380
170	1.810	0.440
195	1.750	0.500
220	1.690	0.560
245	1.630	0.630
270	1.560	0.690
295	1.500	0.750
320	1.440	0.810
345	1.380	0.880
370	1.380	0.880
395	1.250	1.000
420	1.250	1.000
445	1.190	1.060

470	1.130	1.130

Photo documentation of test scenarios are shown in figures 12 to 15:

Figure 12. Initial testing with pultruded slats.

Figure 13. Walking load.

Figure 14. Initial testing of short bridge section.

Figure 15. Standing load.

Conclusion

In conclusion, the ASU team has designed an innovative ladder-bridge that can be utilized by airmen on missions. The device concept is solid, and warrants a promising future. After revising and optimizing the current prototype, such that it more fully aligns with the team's theoretical model, the ladder-bridge could easily replace the bulky units that are employed by soldiers today, and quickly become the preferred method of military field gap navigation.

University Design Challenge 2013-2014

In 2013-14, ASU team was tasked by the Air Force Research Labs (AFRL) with the challenge of designing a heavy lift kit to lift overturned vehicles, aircraft, and building structures in order to help extract trapped personnel and/or equipment. This kit must also be able to be carried by rescue parajumpers to be delivered to the location requiring the necessary operation. After rigorous brainstorming and benchmarking, our team chose to design and prototype a biaxial scissor jack, a scissor jack in which the power screws are perpendicular to each other. With weight being an important issue, we decided to manufacture the jack structure out of 7075-T6 aerospace grade aluminum because of its high strength and low weight. With the solid model completed, we used SolidWorks Simulation analysis to verify that our components would not fail under the heavy loads being applied. With the analysis complete, materials were ordered and the manufacturing processes began. Nearly every component was custom designed and manufactured at our ASU Polytechnic CNC and manual machining labs. With the biaxial scissor jack prototype built, we believed it would be strong enough to withstand the heavy loads and harsh environments it will be subjected to. In this report, you will find detailed drawings of our biaxial scissor jack as well as the simulation analysis reports and other calculations that ensure other components are structurally sound under the heavy loads. In addition to the jack handling the heavy loads, we ensured that the jack can be operated safely using the Operation Risk Assessment provided by the Air Force Research Labs. With everything all said and done, our team feels that we have provided the AFRL with a sufficient design that will meet their needs.

The Air Force Para-rescue Jumpers encounter emergency situations in which crashed vehicles and collapsed buildings trap people. In emergency situations, time is crucial to saving lives. The current system for lifting extremely heavy obstacles is slow and heavy. This is a crucial flaw that costs lives. Air Force Research Labs has requested a new and improved lifting device. They require a faster and more easily deployed system. The team needs to develop a new lifting system that can handle between 45000-55000 pounds, while weighing less than 30 pounds that is easily deployed and operated by untrained personnel.

The Biaxial Scissor Jack

Design Description

The concept of our design was basically taking two ordinary scissor jacks and mounting them perpendicular to each other so that it operated in three dimensions as opposed to only two dimensions. To accomplish this, a gearbox must be used to transfer screw rotation to the other screws so that they turn at the same rate and ultimately provide the lift as the nuts are drawn in towards each other. One of the screws would have a hex nut welded on the end so that the screw can be rotated using a high torque impact wrench as shown in figure 16.

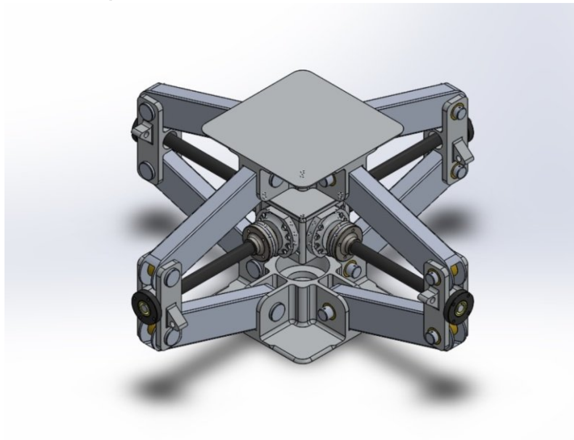


Figure 16: Bi-axial scissor jack

Design Analysis

<u>Screw Analysis</u>

The principle operating mechanism for the bi-axial scissor jack is the four power screws that provide the lift. Our team chose to go with a 1” diameter acme screw made from 4140 steel with a thread pitch of 0.2”. The screw would experience the greatest force the lower the jack gets. We chose to design the jack to start at a 15° angle of inclination. Since the jack was symmetric, each of the 4 sides would experience the same amount of load, so just a single side was analyzed. Assuming the jack is lifting 35,000 lbs (70% of 50,000 lbs due to lifting only partial load) at an angle of inclination of 15° and assuming each plane of the jack’s motion would take half the load, we were able to calculate the maximum magnitude of pull on the screw using the following equation:

$$= \frac{\quad}{2} = \frac{17500}{2 \quad 15°} = 32{,}655$$

Since both nuts experienced the same amount of pull, the total tensile pull on the acme screw is equal to:

$$= 2 \quad = 2(32{,}655) = 65{,}310$$

With this value and knowing the core diameter of the screw (= .7509”), we determined the tensile stress on the screw ( ) using the following equation:

$$= \frac{}{4}(\quad)(\quad)$$

Rearranging this equation to solve for we got the following equation:

$$= \frac{4}{(\quad)} = \frac{4(65310)}{(.7509)} = 147.5$$

In order to calculate the effort to rotate the screw under load, we must solve the following equation:

$$= \quad \tan(\quad + \emptyset)$$

Where $= \frac{}{}$ and $= h$ h (.2") and d_p= pitch diameter of the screw (.8726")

and $\emptyset =$ $(\ )$ $=$ (.16). So,

$$= 65310\tan(\quad \frac{.2}{.8726} + \quad (.16) = 15{,}400$$

With the effort, P, found, we could now calculate the torque, T, required to rotate the screw:

$$= \quad \frac{}{2} = 15400 \frac{.8726"}{2} = 6719 \quad \cdot \quad \frac{1}{12} = 560 \quad \cdot$$

The torque required to rotate the screw creates a shear stress which can be calculated using the following equation:

$$= \frac{16}{\quad} = \frac{16(6719)}{(.7509)} = 80.8$$

Now that we have the tensile stress and shear stress, we calculated the maximum principle tensile stress using the following equation:

$$= \frac{\quad}{2} + \frac{1}{2}\sqrt{\quad + 4\quad} = \frac{147.5}{2} + \frac{1}{2}\sqrt{147.5\quad + 4(80.8)\quad} = 183$$

as well as the maximum shear stress using the following equation:

$$= \frac{1}{2}\sqrt{\quad + 4\quad} = \frac{1}{2}\sqrt{147.5\quad + 4(80.8)\quad} = 109.4$$

Joint Pin Analysis

To help save weight, we decided to make the joint pins from 7075-T6 aluminum. Using the shearing strength of the material, 48 ksi, we were able to calculate the minimum diameter of the pin needed using the following equation:

$$= \frac{\quad}{2} = \frac{\quad}{2\frac{\quad}{4}(\quad)}$$

Therefore,

$$= \sqrt{\frac{\quad}{\frac{\quad}{2}(\quad)}} = \sqrt{\frac{8750}{\frac{\quad}{2}(48000)}} = .34$$

We made the joint pins 1 inch in diameter, so they are more than capable of handling the applied loads.

Gearbox Shaft Analysis

The shafts used 0.75 inches in diameter and are made from 1045 cold drawn steel. The maximum torque would be applied to the shaft when the jack is at a 15° angle of inclination. As the jack lifts, the torque required continues to drop. To find the maximum torsional shear stress in the shaft, we used the following equation:

$$= \frac{\quad @ \quad °}{\quad}$$

where T = torque applied to the shaft, c = radius of the shaft, and J = polar moment of inertia

So,

$$= \frac{(6600\quad \cdot \quad)(.375\quad)}{\frac{(.75\quad)}{32}} = 79.68$$

At a 25° angle of inclination,

$$= \frac{(3852\quad \cdot \quad)(.375\quad)}{\frac{(.75\quad)}{32}} = 46.51$$

As you can see, with just 10° of lift the torque required to lift drops by 2748 in·lb, and thus the max torsional shear stress in the shaft drops by 33.17 ksi. The following chart (table 3) and figure 17 illustrates the different torques required at different angles on inclination :

Load	**Lift angle**	**Tensile load**	**Torque**	**HP @ 10 RPM**	**HP @ 5 RPM**
35000 lb	15	32655.44457	559.7112724	1.065710724	0.532855362
	16	30514.87638	523.0221335	0.995853263	0.497926631

	17	28619.96041	490.5434506	0.934012663	0.467006331
	18	26929.73095	461.5730754	0.878852009	0.439426005
	19	25411.84518	435.5566549	0.829315794	0.414657897
	20	24040.42742	412.0506825	0.784559563	0.392279781
	21	22794.52932	390.6961052	0.743899667	0.371949834
	22	21657.00997	371.1991298	0.706776713	0.353388357
	23	20613.7082	353.3170349	0.672728551	0.336364275
	24	19652.82177	336.8475311	0.641370014	0.320685007
	25	18764.43555	321.6206743	0.612377521	0.306188761
	26	17940.15861	307.4926444	0.585477236	0.292738618
	27	17172.84192	294.340908	0.560435849	0.280217925
	28	16456.35657	282.0604159	0.537053343	0.268526672
	29	15785.41786	270.560589	0.515157252	0.257578626
	30	15155.44457	259.7629055	0.494598068	0.247299034
	31	14562.44547	249.5989564	0.475245538	0.237622769
	32	14002.92713	240.0088642	0.456985652	0.228492826
	33	13473.81843	230.9399905	0.439718185	0.219859092
	34	12972.40847	222.3458706	0.423354666	0.211677333
	35	12496.29506	214.1853311	0.407816701	0.20390835
	36	12043.3418	206.4217546	0.393034567	0.196517284
	37	11611.64219	199.0224635	0.378946046	0.189473023
	38	11199.48928	191.9582011	0.365495432	0.182747716
	39	10805.35012	185.2026927	0.352632697	0.176316349
	40	10427.84394	178.7322719	0.34031278	0.17015639
	41	10065.72356	172.5255625	0.328494978	0.164247489
	42	9717.859505	166.563205	0.317142431	0.158571216
	43	9383.226213	160.8276216	0.306221671	0.153110836
	44	9060.890246	155.3028132	0.295702234	0.147851117
	45	8750	149.9741834	0.285556328	0.142778164
	46	8449.77678	144.8283855	0.27575854	0.13787927
	47	8159.507004	139.8531886	0.266285584	0.133142792
	48	7878.535388	135.0373613	0.257116073	0.128558036
	49	7606.258956	130.3705687	0.248230329	0.124115164
	50	7342.121773	125.843282	0.239610209	0.119805105
	51	7085.61029	121.4466991	0.231238955	0.115619477
	52	6836.249232	117.1726739	0.223101055	0.111550527
	53	6593.597938	113.0136533	0.215182127	0.107591064
	54	6357.24712	108.9626224	0.207468816	0.103734408
	55	6126.815959	105.0130538	0.199948693	0.099974347
	56	5901.949522	101.158864	0.192610175	0.096305088
	57	5682.31644	97.3943735	0.185442448	0.092721224
	58	5467.606829	93.7142708	0.178435398	0.089217699
	59	5257.530416	90.1135807	0.171579552	0.085789776

	60	5051.814855	86.58763518	0.164866023	0.082433011
	61	4850.2042	83.13204736	0.158286457	0.079143229
	62	4652.457527	79.74268784	0.151832993	0.075916496
	63	4458.347683	76.41566323	0.145498216	0.072749108
	64	4267.66015	73.1472967	0.139275127	0.069637564
	65	4080.192009	69.93411026	0.133157103	0.066578551
	66	3895.750996	66.77280852	0.127137868	0.063568934
	67	3714.154642	63.66026395	0.12121147	0.060605735
	68	3535.229476	60.5935033	0.115372245	0.057686123
	69	3358.810307	57.5696952	0.109614804	0.054807402
	70	3184.73955	54.58613868	0.103934004	0.051967002
	71	3012.866616	51.64025264	0.098324929	0.049162464
	72	2843.047342	48.72956612	0.092782875	0.046391438
	73	2675.143463	45.8517093	0.087303331	0.043651665
	74	2509.022125	43.00440508	0.081881959	0.04094098
	75	2344.555434	40.18546134	0.076514587	0.038257294

Table 3: Load and power requirements

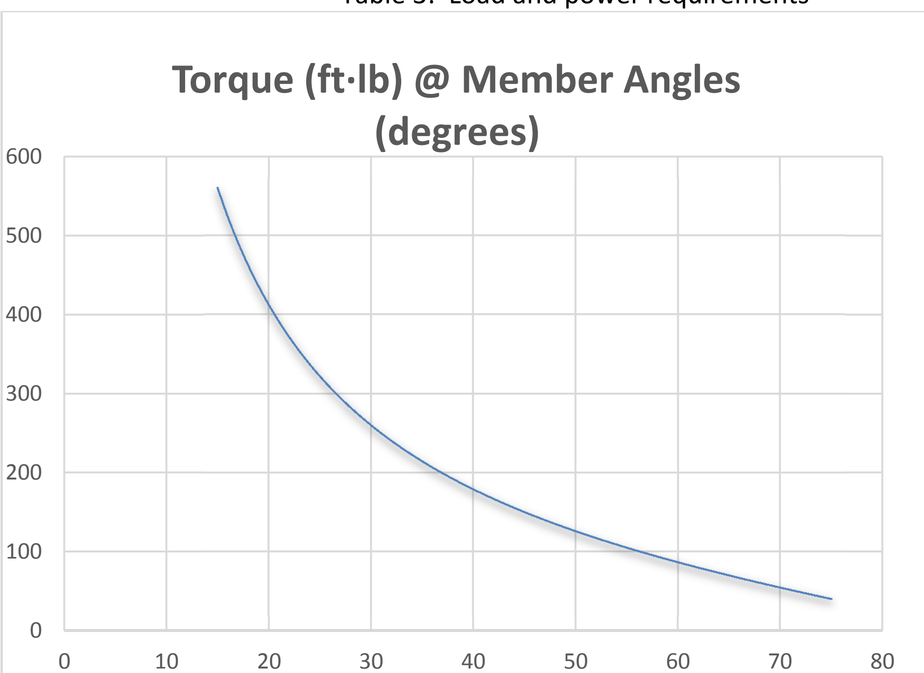


Figure 17: Lift Angle v/s Torque

Miter Gear Analysis

The gears used were three Browning miter gears (part # YSM10F20H3/4). They feature 20 teeth (N), a diametral pitch ( ) of 10, a pitch diameter (d) of 2.00 inches, .75 inch bore, a face width (F) of .44 inches, pressure angle (Ø) of 20°, and a pitch cone angle ( ) of 45°. With these values, we were able to calculate the tangential, radial, and axial forces acting upon the gears using the following set of equations:

Tangential Force:

$$= \frac{}{} = \frac{}{\frac{}{2} - \frac{}{2}} = \frac{6720}{\frac{2}{2} - \frac{.44}{2} \quad 45} = 7962$$

Radial Force:

$$= \quad \emptyset \quad = 7962 \tan(20) \cos(45) = 2049$$

Axial Force:

$$= \quad \emptyset \quad = 7962 \tan(20) \sin(45) = 2049$$

<u>SolidWorks Simulation Analysis</u>

After developing solid models of our design, some of the first tasks performed were running Finite Element Analysis (FEA) simulations using SolidWorks to see if the individual components can withstand the forces applied. Because our biaxial scissor jack is completely symmetric with multiples of the same components, we were able to analyze just one of each type of component. In order to do so, we took a maximum lifting load of 35,000 lbs and divided that by four, 8750 lb, to be applied to each component. More detailed simulation reports will be present in the appendices.

The first component we ran simulations on was one of the member arms. We applied a bearing load on the surfaces of the holes on one side and tested it for static stress, displacement and strain as well as buckling displacement. The member experienced a stress value of 24.68 ksi, which is well within the materials 73.24 ksi yield strength as shown figures 17 below.

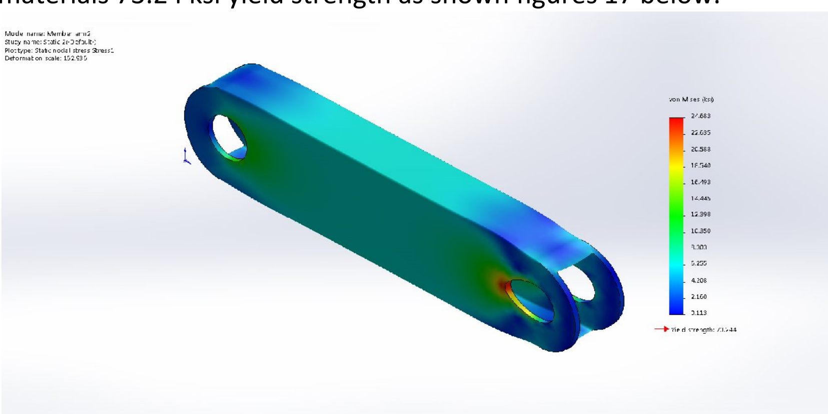


Figure 17: von Mises stress under static bearing load of 8750 lb.

Under the static bearing load, the member also experienced a URES displacement (= $\sqrt{\Delta \quad + \Delta \quad + \Delta \quad}$) of .0092 in. and a strain of .00192 as shown in figures 18-19 below.

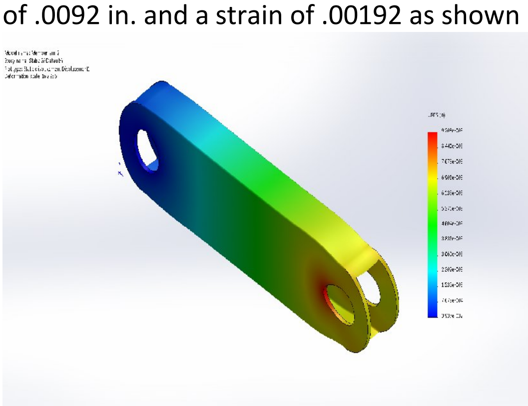


Figure 18: URES Displacement under static bearing load of 8750 lb.

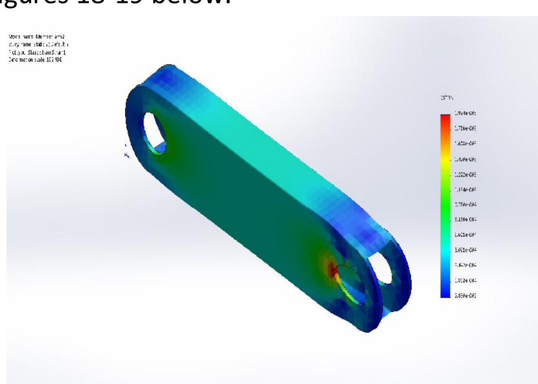


Figure 19: Strain under static bearing load of 8750 lb

The member arm also experienced a buckling displacement of .1243 in. as pictured in figure 20 below.

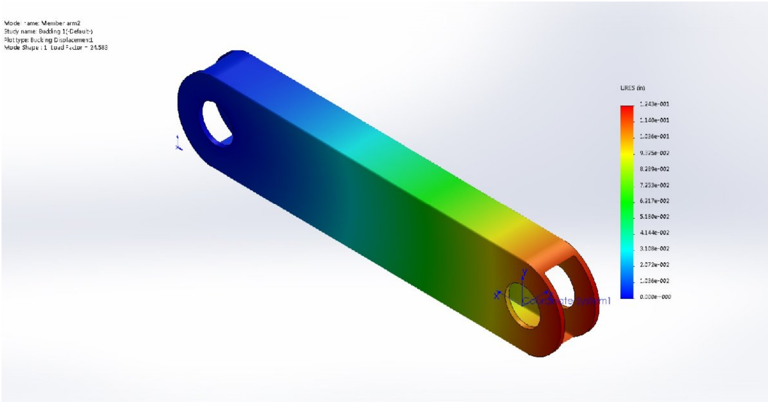


Figure 20: Buckling URES displacement on member arm under 8750 lb load.

The next component we ran simulations on was the nut housing. A bearing load of 8750 lb was applied to both top and bottom holes at the jack's lowest angle of inclination, 15°, to determine the parts stress, displacement, and strain. The nut housing experienced a maximum von Mises stress of 37.5 ksi, which is well within the materials yield strength of 73.24 ksi as shown in the figure 21 below.

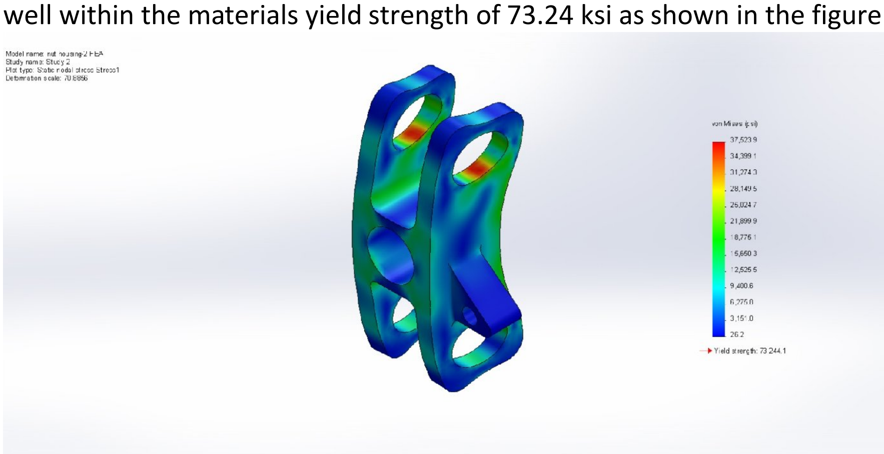
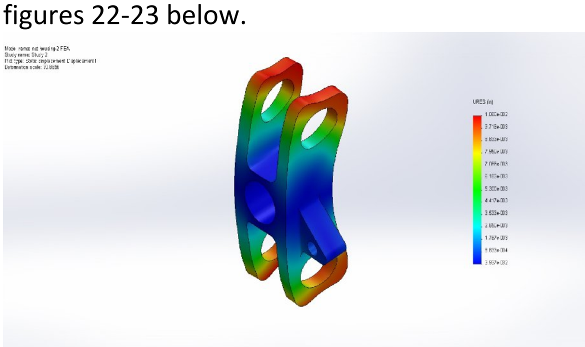


Figure 21: von Mises stress on nut housing under static bearing load of 8750 lb

The nut housing also experienced a displacement of .0106 in and a strain of .0028. as shown in the figures 22-23 below.



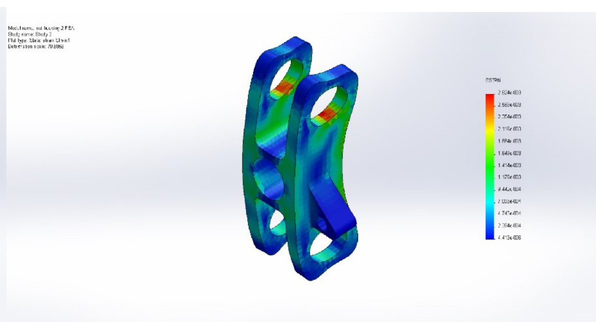


Figure 22: Displacement on nut housing under 8750 lb bearing load   Figure 23: Strain on nut housing under 8750 lb bearing load

The last component that we ran simulations on was the base/top plate, which we believe to be the strongest of the components in the jack system. Each side of the plate was subjected to a bearing load of 8750 lb. As shown below, the maximum von Mises stress was 12.28 ksi, well below our material's yield strength.

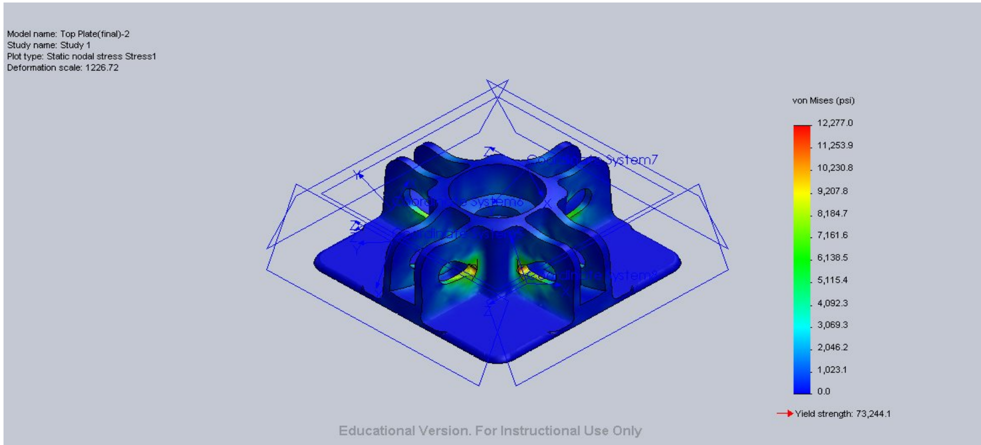


Figure 23: von Mises stress under 8750 lb load on all four sides.

The base/top plate also experiences a URES resultant displacement of .00105" and a strain of .0009 as illustrated below.

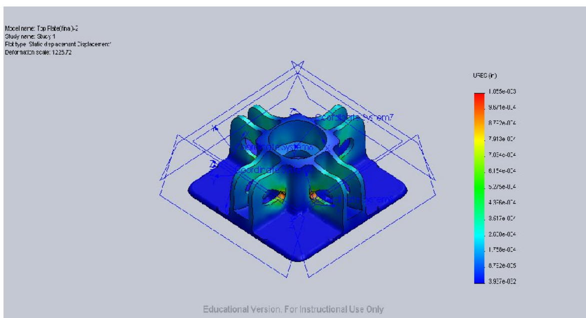
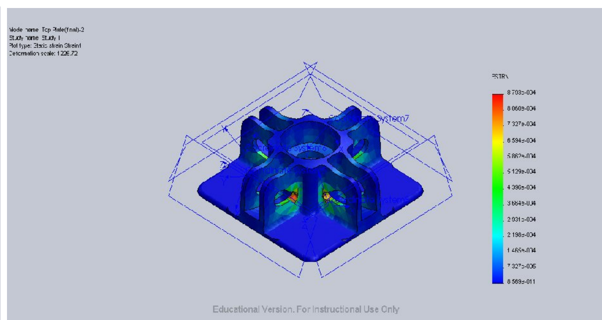



Figure 24: URES displacement on plate under bearing load   Figure 25: Strain on plate under bearing load

Analyzing the results of all the SolidWorks simulations, our team believes that each of the components are more than capable of withstanding at least 35,000 lbs of force applied to the jack.

Manufacturing the Prototype

Almost all of the parts of this design were manufactured in our CNC and manual machining labs on the ASU Polytechnic campus out of stock material, shown in the figure below:

Conclusion

After countless hours of hard work and dedication, our team feels that we have come up with a plausible solution for the AFRL's problem in lifting heavy equipment. We brainstormed several ideas until we came up with the best and most innovative design. Through careful analysis and soon to be testing with the limited time remaining, we feel that our design will fulfill the critical design requirements and safely lift and operate in the conditions demanded.